REPORT DOCUMENTATION PAGE	Form Approved OMB No. 0704-0188

1. REPORT DATE *(DD-MM-YYYY)*	2. REPORT TYPE	3. DATES COVERED *(From - To)*
28-02-2003	A005 DATA ITEM	01-04-1998 TO 28-02-2003

4. TITLE AND SUBTITLE FINAL REPORT for Contract DAAE07-98-C-M012 (PART 1 OF 3) -Progress Report of LMTV Flywheel Housing & Driveline Failure Investigation	**5a. CONTRACT NUMBER** DAAE07-98-C-M012 **5b. GRANT NUMBER** **5c. PROGRAM ELEMENT NUMBER**
6. AUTHOR(S) HAYES HOBOLTH	**5d. PROJECT NUMBER** **5e. TASK NUMBER** **5f. WORK UNIT NUMBER**
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) MICHIGAN SCIENTIFIC CORPORATION 730 BELLEVUE AVENUE MILFORD, MI 48381	**8. PERFORMING ORGANIZATION REPORT NUMBER** A005
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) UNITED STATES ARMY- TACOM WARREN, MI 48397-5000 JAMES YAKEL, COTR	**10. SPONSOR/MONITOR'S ACRONYM(S)** N/A **11. SPONSOR/MONITOR'S REPORT NUMBER(S)** N/A

12. DISTRIBUTION/AVAILABILITY STATEMENT

DISTRIBUTION STATEMENT A

20030710 036

13. SUPPLEMENTARY NOTES

TEXT DOCUMENT IN PAPER COPY

14. ABSTRACT

THIS PROGRESS REPORT DEFINES ROOT CAUSES OF THE DRIVELINE AND POWERPLANT COMPONENT FAILURES OF LMTV VEHICLES. THESE FAILURES HAVE RESULTED IN A SAFETY OF USE MESSAGE RESTRICTING OPERATING SPEEDS TO 30 MPH MAXIMUM. POSSIBLE CORRECTIVE ACTIONS ARE DISCUSSED.

15. SUBJECT TERMS

FMTV, LMTV, DRIVESHAFT, CARDAN, CV, CONSTANT VELOCITY, FLYWHEEL HOUSING, BELL HOUSING, CRITICAL SPEED, DRIVELINE, DRIVELINE DYNAMICS, DRIVELINE MODEL, POWERPACK, POWERPACK DYNAMICS, TRANSFER CASE TRANSDUCER, DRIVESHAFT BALANCE, DADS MODEL.

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON / 19b. TELEPHONE NUMBER *(Include area code)*
UNCLASS	UNCLASS	UNCLASS	UNCLASS	36	HAYES HOBOLTH 248-685-3939

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PROGRESS REPORT OF LMTV FLYWHEEL HOUSING AND DRIVELINE FAILURE INVESTIGATION

Contract No. DAAE07-98-C-M012

Hayes Hobolth
17 August 1998

PR1.DOC

INTRODUCTION: Low mileage failures of flywheel housings and related driveline components became unacceptable by mid 1997. A Safety of Use Message was issued restricting speeds of operation for both LMTV and MTV, which accelerated the search for solutions. Caterpillar issued statements indicating that unbalance in the driveshafts was the primary issue and that the strain levels in the flywheel housings would be acceptable if unbalance was kept low. A contract was awarded to Michigan Scientific Corporation in April of 1998, part of which was to identify the cause(s) and recommend corrective actions for the deficiencies noted. Mr. Gary Schultz was named as Contracting Officer's Technical Representative, and later, Mr. Donald Dismang as his Alternate.

PHASE I: A Start of Work meeting was held at Michigan Scientific Corporation's Milford, Michigan facility on April 21, 1998, and was attended by Cpt. Brent Thomas and Mr. James Lim. A significant investigation and information gathering had already been done by Mr. Hayes Hobolth as a consultant to the PM-FMTV as an employee of SAIC. A summary statement of the cause of the problems was dated April 14, 1998 and is based on analysis of data presented by Caterpillar, US Army ATC, EG &G Automotive Research (contracted by Tactical Vehicle Systems, the prime contractor) and numerous field trips, and examination of failed parts. In short, the cause of the problems was attributed to a resonant configuration involving the two main masses of the engine and the transmission/transfer case assembly separated by an elastic flywheel housing which causes the system to have a resonant response at 42-48 Hz. This frequency is also that of the first order of the driveshaft speed at about 47 mph, where the majority of the on-road time is spent. When the vehicle is driven at its maximum speed of 58-60 mph, the first order of the engine speed also excites the resonant response, by virtue of the overdrive gear ratio. Depending on the particular vehicle, either source may be dominant at either speed. It was agreed at the Sealy Texas PM review meeting that the resonant system must be tuned out of the driving range, if possible.

Since the primary failures were cracks in the flywheel housing, and it was believed to be the frequency setting spring in the system, MSC, with TACOM approval, ordered experimental castings of higher strength nodular iron and with increased section as a diagnostic tool. The material had a specified 80,000 psi ultimate strength compared to the Caterpillar specification of 35, 200 psi.

The preliminary results of tests on the first housing produced, were presented to TACOM at Aberdeen Proving Ground on July 15, and again at Caterpillar on July 28,'98. Strains were well below design limits for that material, and the apparent frequency of the vertical bending mode was 65-68 Hz, which was consistent with the measured 360-400 percent static stiffness increase measured at the MSC laboratory. This investigation was primarily diagnostic in its intent, but now is considered a possible corrective action.

The driveline unbalance includes the internal rotating parts in the transfer case and a significant unbalance of the torque converter and engine cooling fan, both of which run at engine speed. Some early LMTVs had a fan design which was found to contribute as much as 12 ounce inches of unbalance*.

An important relationship is shown in Figure 1, which is a mode shape of the vertical bending mode of the resonant response at its peak excited by the an unbalance at the transfer case output. The dynamic first order vertical velocity was probed at 10 inch increments along the length of the powertrain and is plotted on a side view. It clearly shows that the elastic energy is stored in the deformation of the flywheel housing.

* Determined by EG &G and presented at Sealy, Tx

PHASE II- Identify corrective actions: Diagnostic tests often point to corrective actions as is demonstrated above. A flywheel housing having stiffness and strength properties comparable to the experimental MSC flywheel housing would correct the flywheel housing failure problem.

Tests were also performed in an attempt to develop a tuned-mass damper. The first attempt used a 94 lb. Mass and showed about a 30 percent reduction in flywheel housing strain. A less massive version was tested and failed to show any significant benefit. This approach was abandoned .

MSC attended several meetings where results of a Cradle stiffener, and an added center mount were discussed. While both of these approaches demonstrate some benefit, neither is currently recommended.

DRIVELINE INFLUENCES: In addition to the unbalance as manufactured, the driveshafts are susceptible to the following:

1. Runout due to assembly. The U-joint seats in the Transfer case and axle yokes allow off center installation, which causes the entire mass of the driveshaft to orbit the center of rotation.
2. Bending of the 3 ½ inch driveshaft tube.
3. Bending of the slender portion of the slip yoke.
4. Deformation and **wear** of the plastic coating of the slip-yoke spline. Data being accumulated by ATC indicates a steady increase of the force with miles.

All of these contribute to the cumulative total unbalance force vector which is often hundreds of pounds and clearly challenges the fatigue strength of the production flywheel housing. This underscores the necessity of tuning the primary resonance out of the driving speed range.

Future Progress Reports will deal more with driveshaft diagnoses and corrective actions.

Attachments:

a. Memorandum for record- April 14, 1998.
b. Start of Work Meeting – handout
c. Preliminary Results of Ductile Iron Flywheel Housing Tests.
d. Figure 1. Vertical Bending Mode Shape.

Hayes M. Hobolth

April 14, 1998

MEMORANDUM FOR RECORD

Subject: Summary statement of root cause of Flywheel housing and driveshaft failures LMTV

To: Cpt. Brent Thomas, Mr. James Lim, Gary Schultz

Gentlemen:

To summarize the status of the knowledge I have been able to gain on the above open issue, please consider the following:

1. The root cause of the problems is cumulative unbalance of components which rotate at between 40 and 60 times per second. The relative contributions of the components and the sensitivities of the system to these are to be quantified by Michigan Scientific Corporation (MSC). The components include but are not limited to the driveshafts, transmission and transfer case, engine and fan. As much as is feasible, they will be quantified separately and in combination in terms of response and measured strains.

2. The principle mitigating factor is the distribution of mass and stiffness of the engine, flywheel housing and the transmission/transfer case assembly which forms a spring-mass system having one or more natural frequencies in this range. The dynamic behavior of the system is being quantified by TVS (E G & G) and Caterpillar. Experimental changes to the system will be created and/or evaluated by . E G & G, MSC and others in terms of the response of the system and likelihood of success.

Cooperation and sharing of knowledge between parallel investigations is understood as imperative for an early solution.

Hayes Hobolth

Hayes Hobolth

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PRELIMINARY RESULTS OF DUCTILE IRON FLYWHEEL HOUSING TESTS

Hayes Hobolth
July 15, 1998

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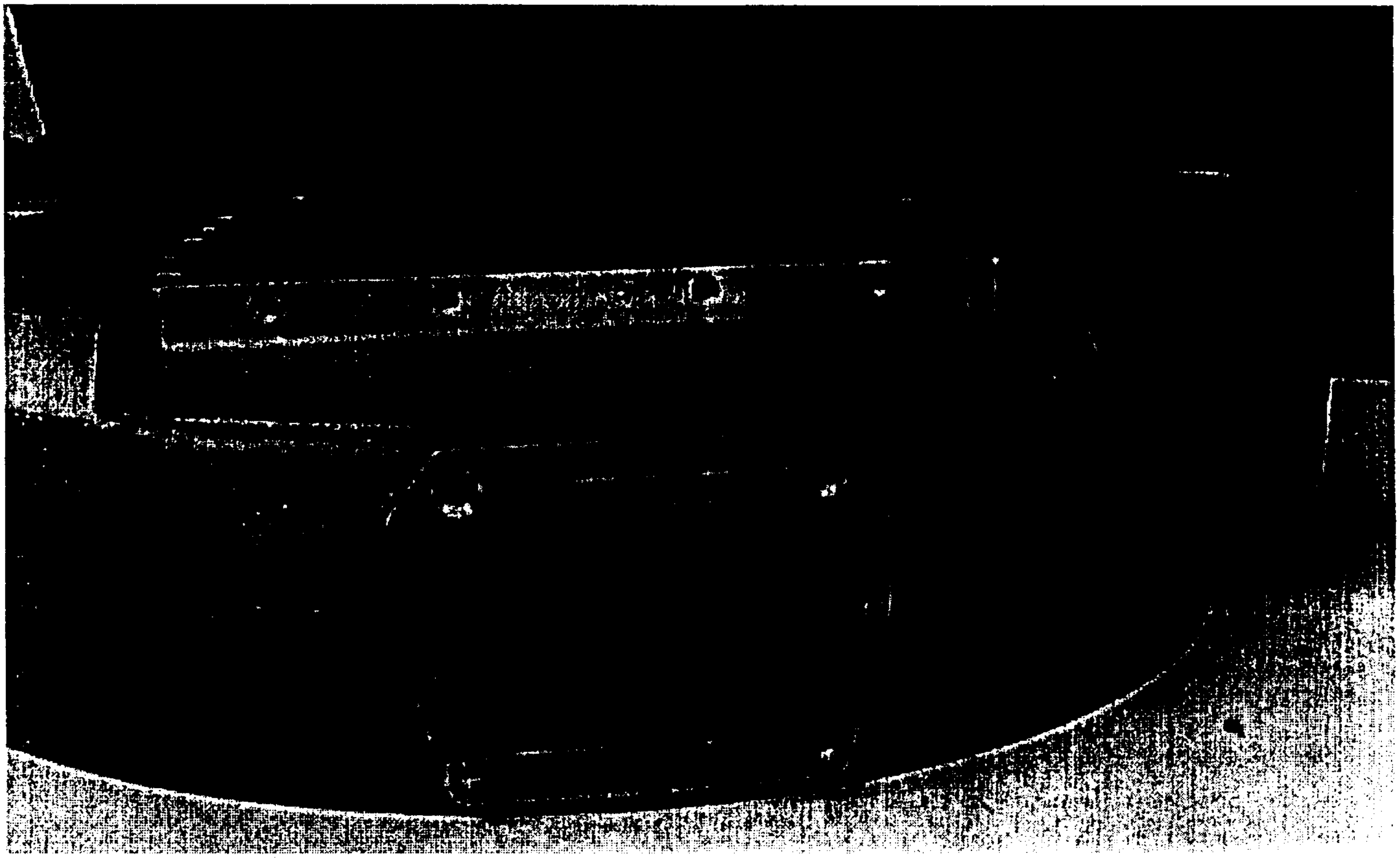
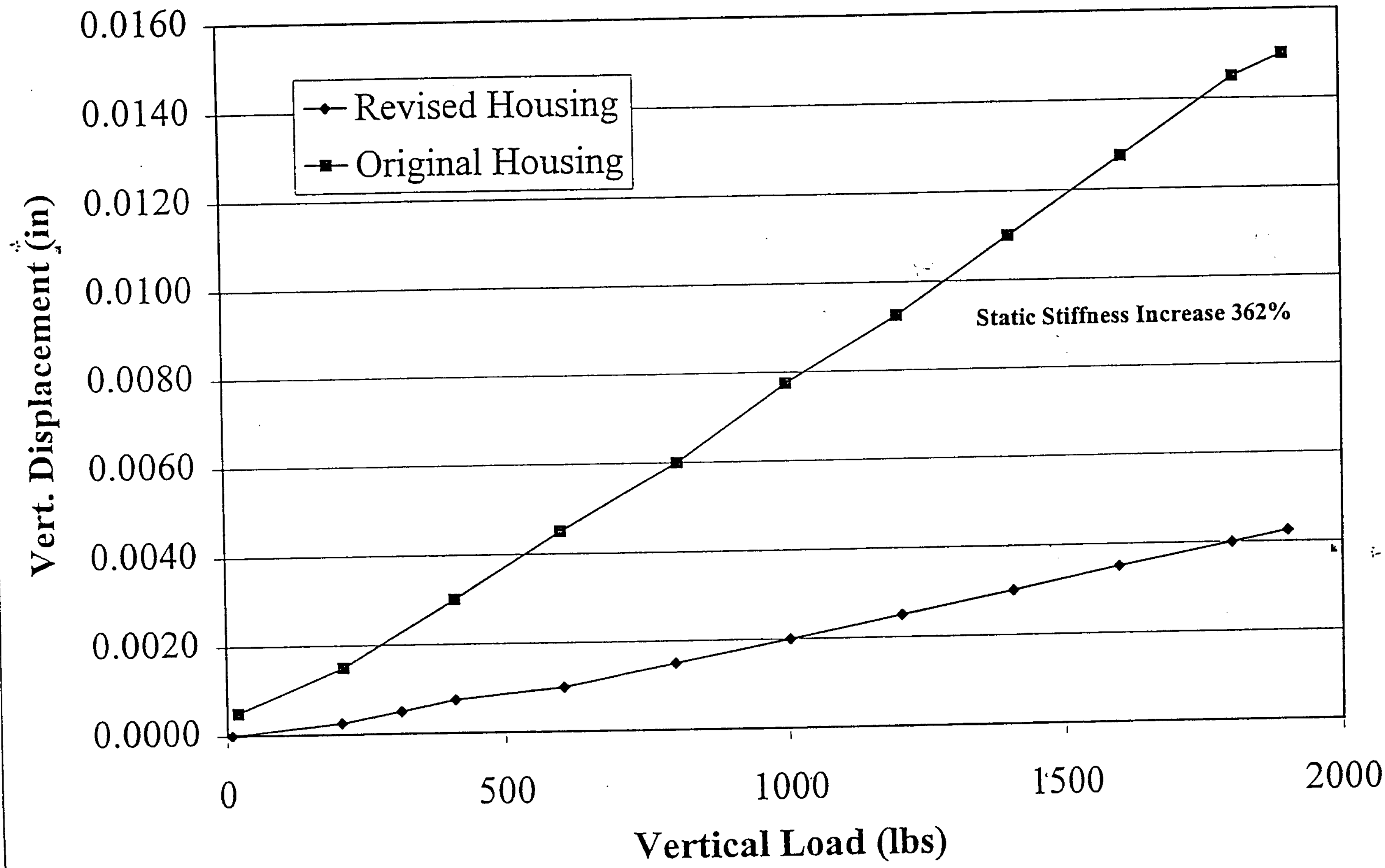


Figure 1

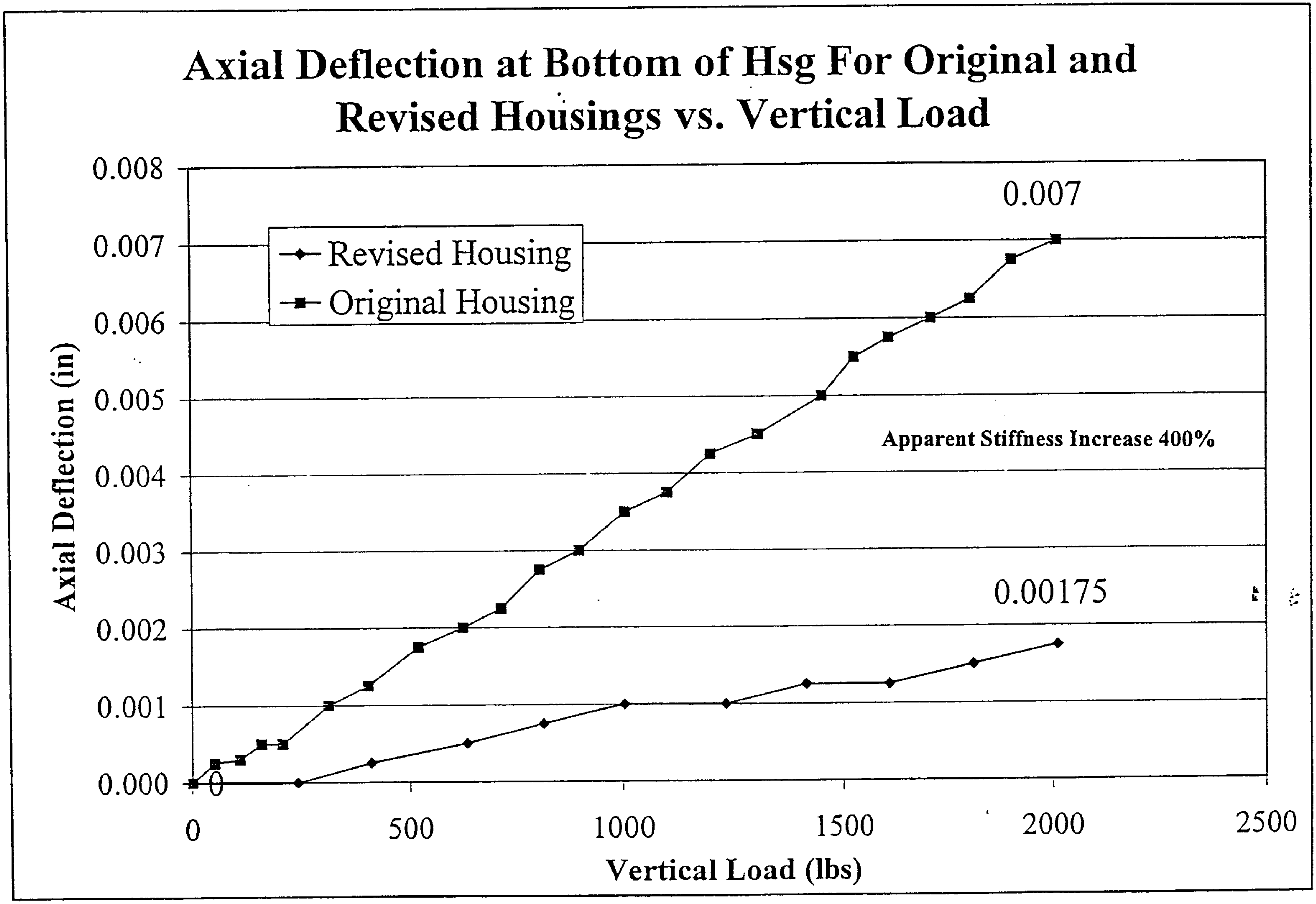
EXPERIMENTAL STIFFENED FLYWHEEL HOUSING

Vertical Displacement vs. Vertical Load at the Flywheel Housing - Original and Revised Housings



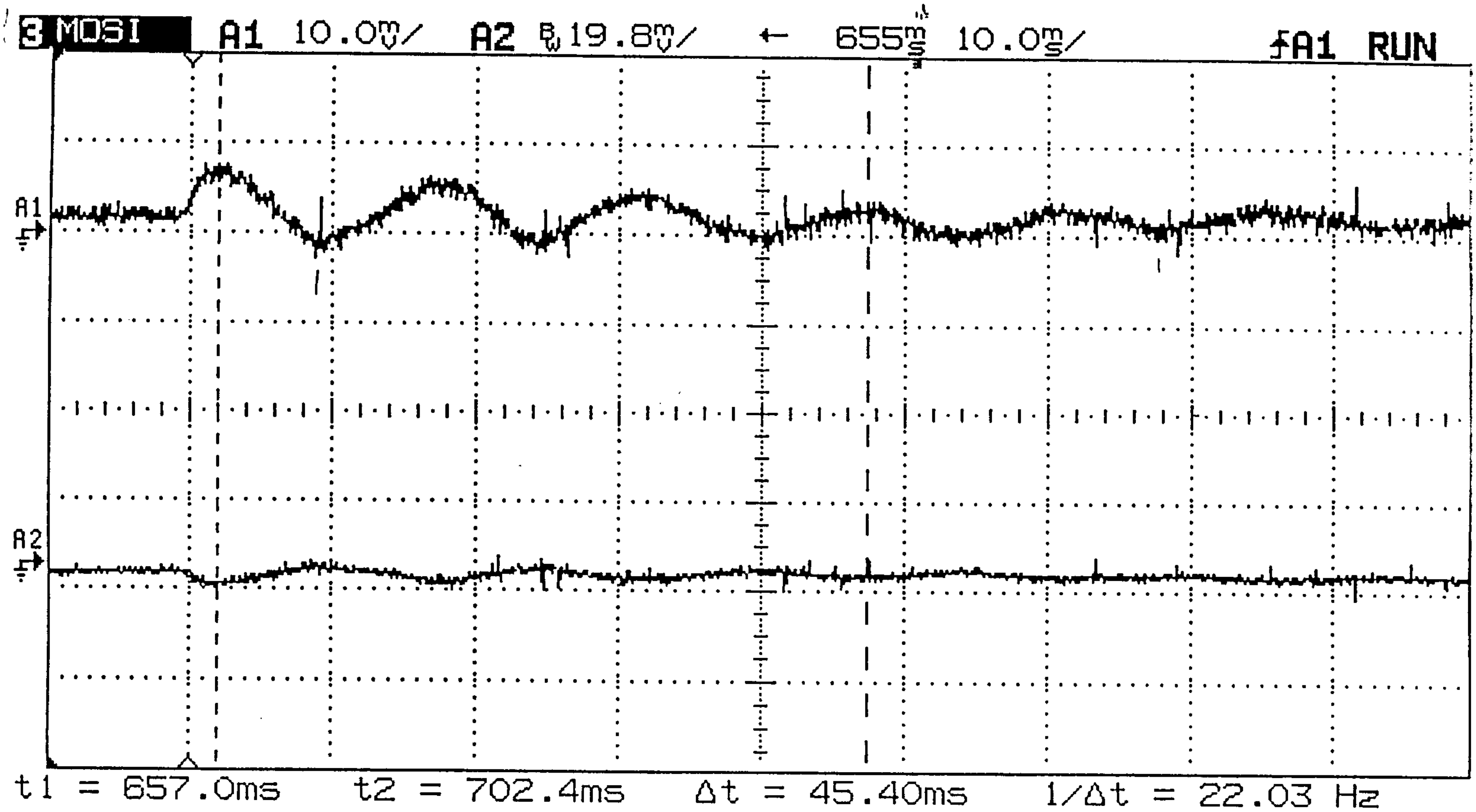
R0-Test#5, R1-Test#2

July 9, 1998



Axial Deflection at Bottom of Hsg For Original and Revised Housings vs. Vertical Load
Revised Housing
Original Housing
0.007
0.00175
0
Apparent Stiffness Increase 400%
Axial Deflection (in)
0.008
0.007
0.006
0.005
0.004
0.003
0.002
0.001
0.000
0
500
1000
1500
2000
2500
Vertical Load (lbs)

FMTV7-8.PCX HP54645D <1> 98-07-08 16:12:30PM



Impact Test of Natural Frequency 65-68 Hz (Original – 47.6 Hz)

Preliminary Results of First Mode Vertical Bending

- Impact test agrees with Caterpillar projection for ductile iron and increased section.
- Speed sweeps do not show a clearly defined resonance.
- Probing for the characteristic first mode vertical bending suggest that the new bending mode may be critically damped

Maximum Recorded Flywheel Housing Strains at Gage Locations

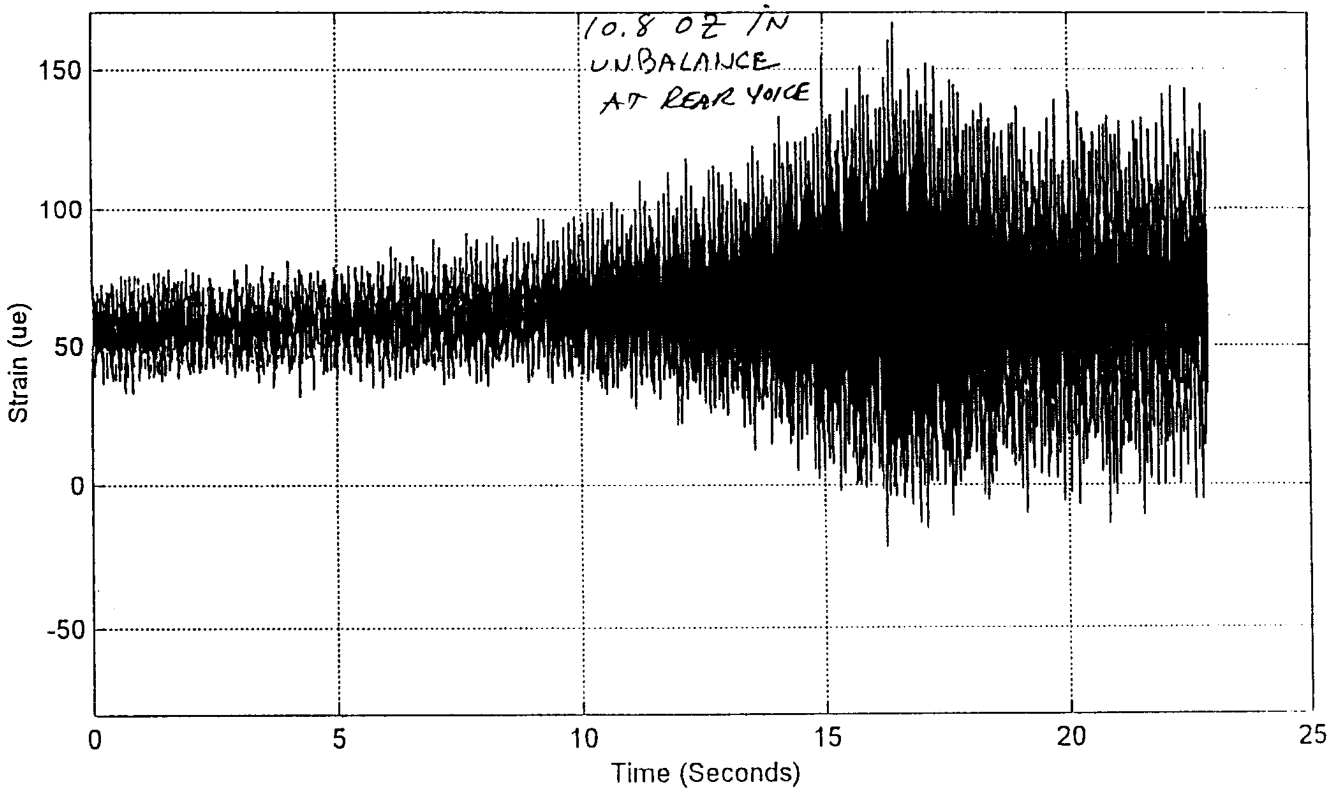
Old Housing - Maximum Strain (Right Side Lower Gages)

Cases	Maximum Peak to Peak Strains (uE)
Neutral	970
6th gear - no imbalance	920
7th gear - no imbalance	875
6th gear - 2.7 in-oz imbalance	1396
7th gear - 2.7 in-oz imbalance	1057
6th gear - 5.9 in-oz imbalance	1293
7th gear - 5.9 in-oz imbalance	1037
7th gear - 10.8 in-oz imbalance	1645
7th gear - 25.5 in-oz imbalance	2300

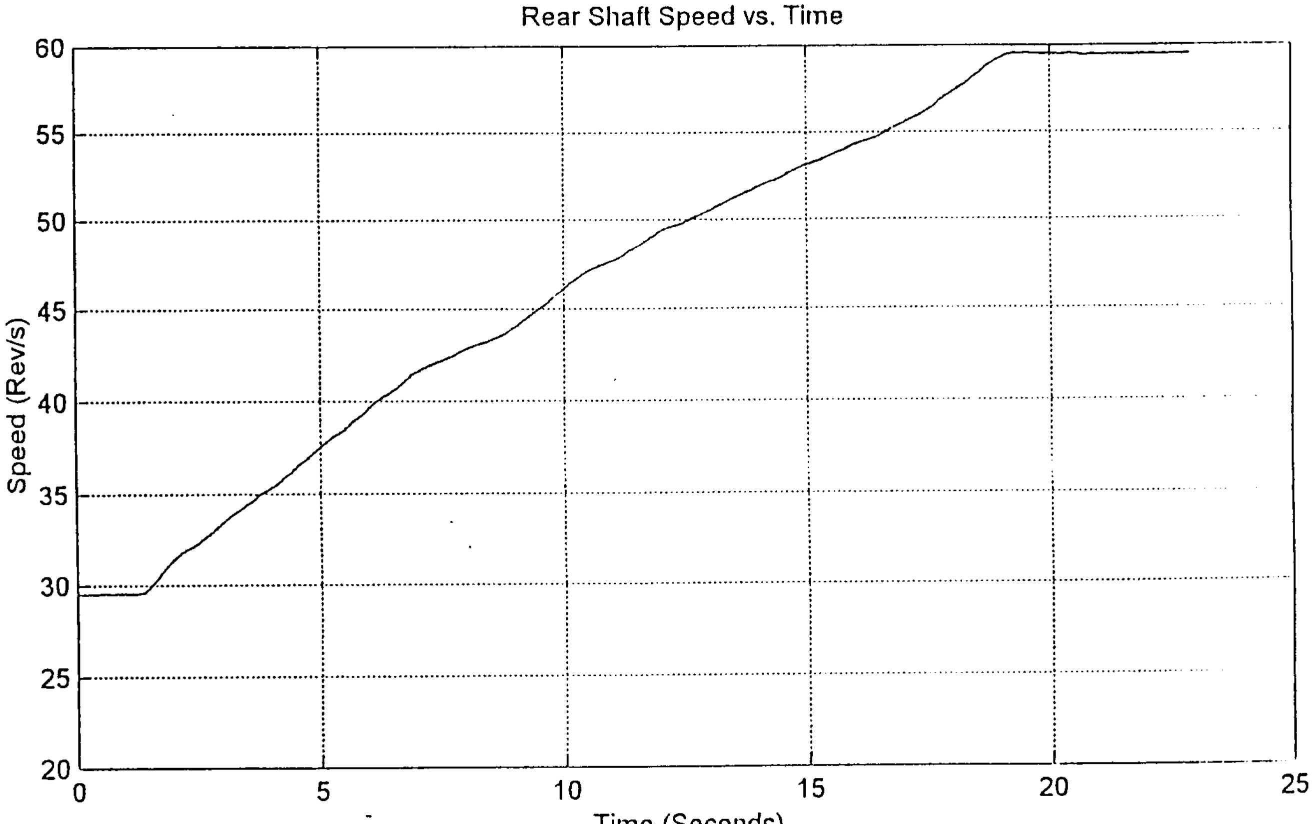
New Housing - Maximum Strain (Left Side Upper Gages)

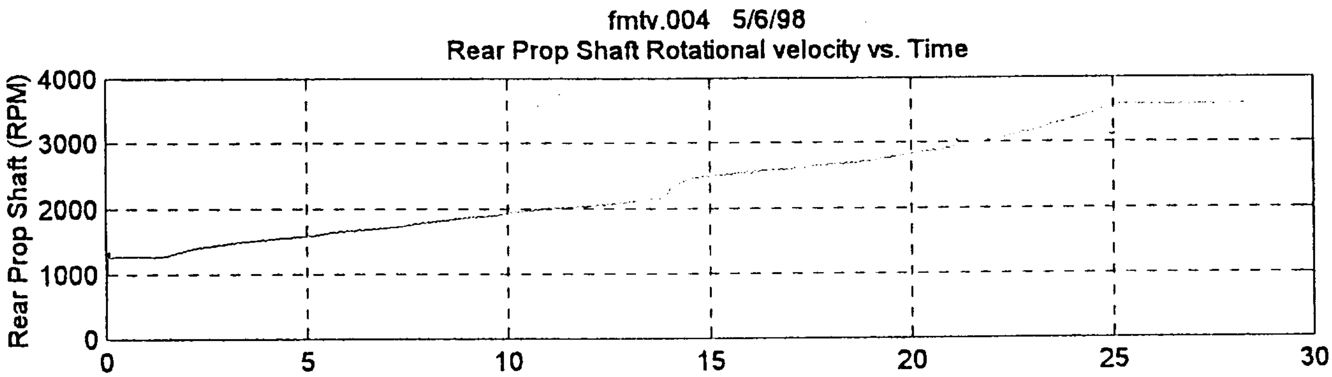
Cases	Maximum Peak to Peak Strains (uE)
Neutral	65
6th gear - no imbalance	92
7th gear - no imbalance	133
6th gear - 2.7 in-oz imbalance	93
7th gear - 2.7 in-oz imbalance	122
6th gear - 5.9 in-oz imbalance	87
7th gear - 5.9 in-oz imbalance	121
6th gear - 8.5 in-oz imbalance	125
7th gear - 8.5 in-oz imbalance	137
6th gear - 10.8 in-oz imbalance	119
7th gear - 10.8 in-oz imbalance	169

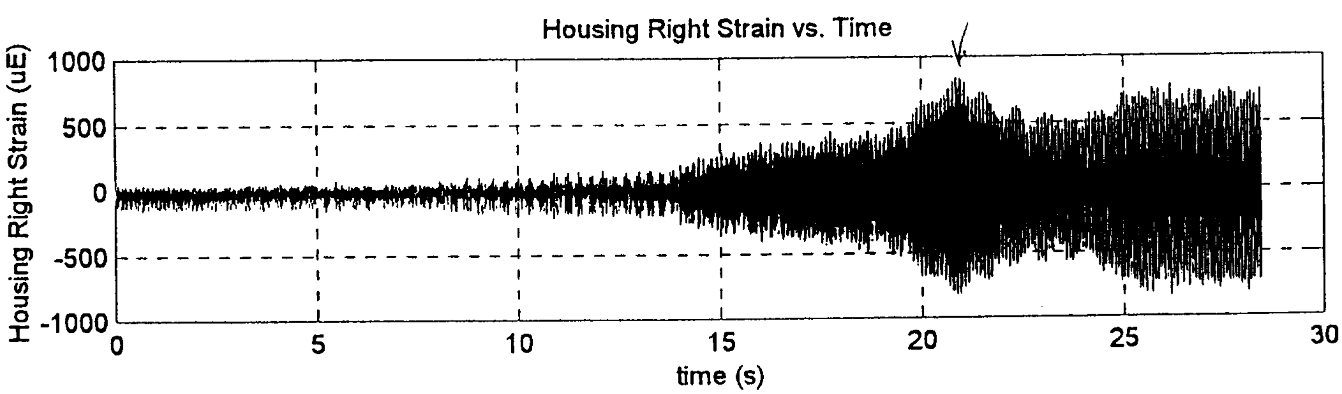
Dfmtv2k.004 7/10/98
Upper Left Strain vs. Time



Rear Shaft Speed vs. Time







10.8 OZ IN UNBALANCE AT
REAR YOKE

Post-it® Fax Note 7671	Date 7-9-98	# of pages ▸ 3
To HAYES HOLBOLTH	From MARK PADESKY	
Co./Dept. TACOM	Co. CATERPILLAR	
Phone # 810-574-8838	Phone # (309)578-2806	
Fax # 810-574-8835	Fax # (309)578 3739	

Thomas J. Majewski

06/15/98 09:54 AM

To: Mark E. Padesky, Thomas D. Keeting
Subject: Lab Paper MS00002018 - HOUSING-FLYWHEEL
Retain Until: 07/15/98 Retention Category: G90

This test bar meets ASTM 80-60-03 ductile iron. As to which spec of Caterpillar it meets. It fits better with our 1E596 which is a ASTM 80-55-06 modified. The tensile strength of the test bar is 96.4 ksi. Mossville met lab does not have extensometer big enough to run on a .505 tensile bar so we did not obtain the yield or elongation. Looking at the microstructure, it is almost all pearlite (85%) similar to a typical 1E0596 microstructure. 1E596 is a higher tensile strength grade than 1E356 which is a more ferritic grade with a tensile of 60.2 ksi. Chemistry meets either spec. If you other question please feel free to get a hold of me.

Tom Majewski

86269

---------------------- Forwarded by Thomas J. Majewski/OC/Caterpillar on 06/15/98 09:19 AM ----------------------

PEPD METLAB

06/15/98 08:41 AM

To: Thomas J. Majewski
Retain Until: 07/15/98 Retention Category: G90

Caterpillar - Mossville Met Lab

PEPD
Mossville Met Lab
0000000000
Mossville, IL 61552

Lab Paper: MS00002018
Lab Code: B03C - DEVELOPMENT - MISCELLANEOUS
Part Number: 1265874
Part Description: HOUSING-FLYWHEEL
Sample Qty: 1
Submit Date: 6/11/98
Due Date: 6/12/98
Completion Date: 6/12/98
Material 1E Spec.: 1E356

Submitter: MAJEWSKI, THOMAS
Phone: ( ) 8-6269
Charge Code: 79-83653
Disposition Required: Yes
Special Filed: No
R and D: No

Dispositioned As INFORMATION ONLY By LES BRAKER

Lab Paper Comments

MICHIGAN SCIENTIFIC
corporation

START OF WORK MEETING

FMTV FLYWHEEL

INVESTIGATION

April 21, 1998

Hypothesis:

1. The vehicle is OK if built to the design tolerances.

   Test:
   - Locate out of spec parts
   - Replace or repair
   - Stack worst case probability
   - Retest
   - Determine limit unbalance

2. Design included a resonant configuration that will not tolerate the probable shaker magnitudes without failure.

   Test:
   - Establish mode
   - Develop means of improving the stiffness/mass ratio
   - Test for strain at Flywheel housing and accelerations
   - Includes experimental Flywheel housing bracing and/or dynamic absorber
   - Use limit stack unbalance and higher
   - Determine margin for a robust design
   - Check for "New Problems"

This rather paradoxical statement is not quite so important as it sounds. In the first place, the bad effect of damping is not great and can be easily offset by making the springs somewhat weaker, *i.e.*, by moving somewhat more to the right in Fig. 52. On the other hand, though it is not our *intention* to run at the resonance point $\omega/\omega_n = 1$, this unfortunately may sometimes occur, and *then* the presence of damping is highly desirable. Thus in spite of the dictum of Fig. 52, some damping in the springs generally is of advantage.

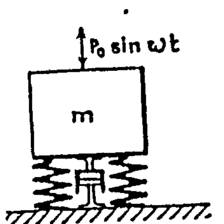


FIG. 51.—A spring support with damping.

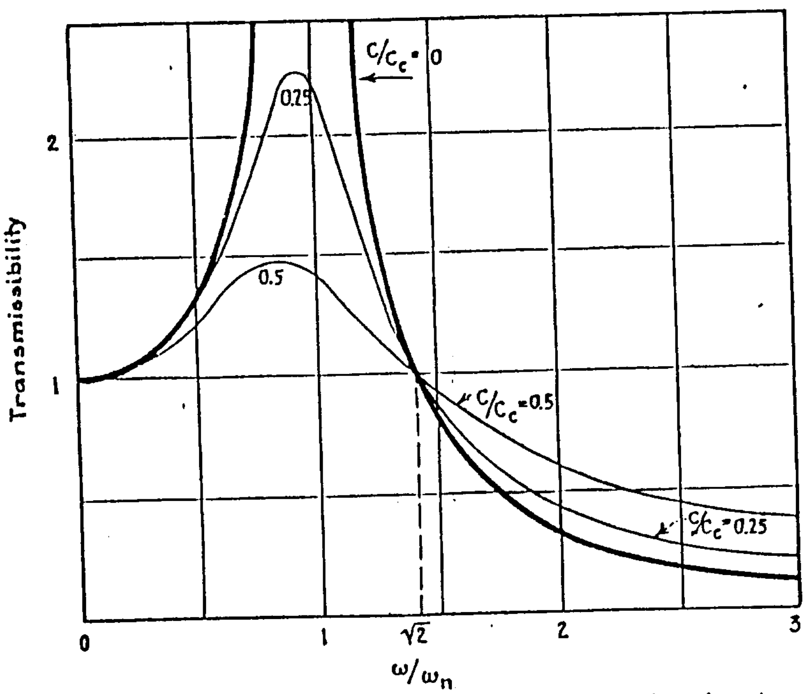


FIG. 52.—Showing that damping in the spring support is advantageous for $\omega < \omega_n\sqrt{2}$, but is detrimental for $\omega > \omega_n\sqrt{2}$.

20. Application to Single-phase Electrical Machinery.—Practical cases of isolation by means of springs occur in many machines. The main field of application, however, lies in apparatus which is inherently unbalanced or inherently has a non-uniform torque. Among the latter, single-phase electric generators or motors and internal-combustion engines are the most important.

First, single-phase machines are to be discussed. As is well known, the torque in any electric machine is caused by the pull of the magnetic field on current-carrying conductors. The magnetic field itself is caused by a current flowing through the field coils. If the machine is operated by single-phase alternating

DEN HARTOG
VIBRATIONS - 1940

Exciters

Sources:	Frequencies:
Wheel out of round	1/rev, 2/rev
Tire nonuniformity	1/tread lug…
Engine	1,2,3,6
Fan	
Accessories	
Transmission	
Converter	1/rev input
Clutches	Individual speeds
T-case –gears, output yokes	Driveshaft speed
Driveshafts	1/rev
U-Joints	2/rev
Road Inputs	Varied & random

- A responding system will "phase-lock" with whatever is exciting it.
- The responding body may vibrate at the principal frequency of the shaker and also at harmonics of that frequency.

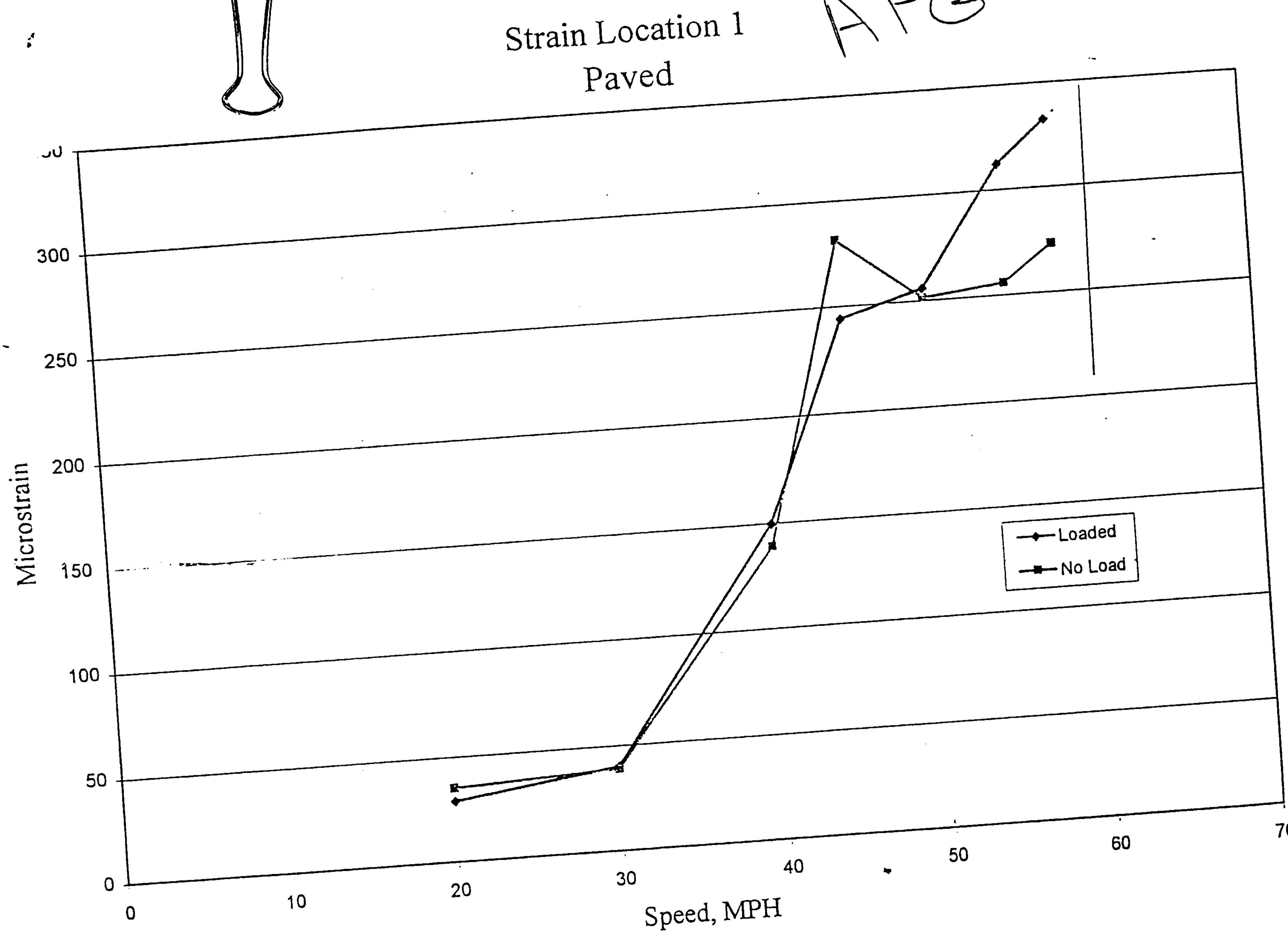
  Example: Cat data showing 2nd and 3rd orders.

- The responding body will not respond at sub-harmonic frequencies.

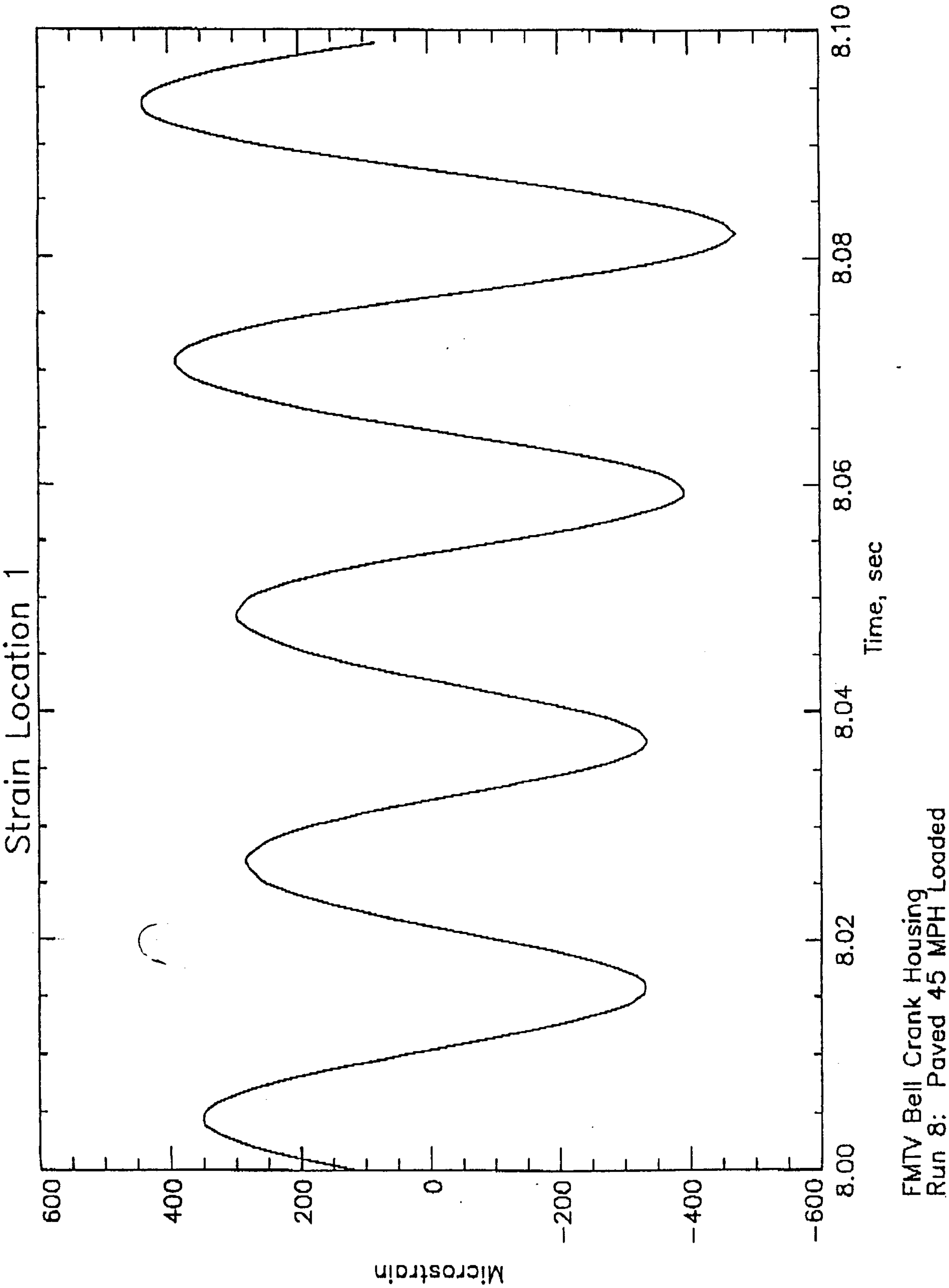
  Example: A U-joint will not cause first order response.

- We see clear evidence of both first order driveline and first order engine frequencies in somewhat similar magnitudes.

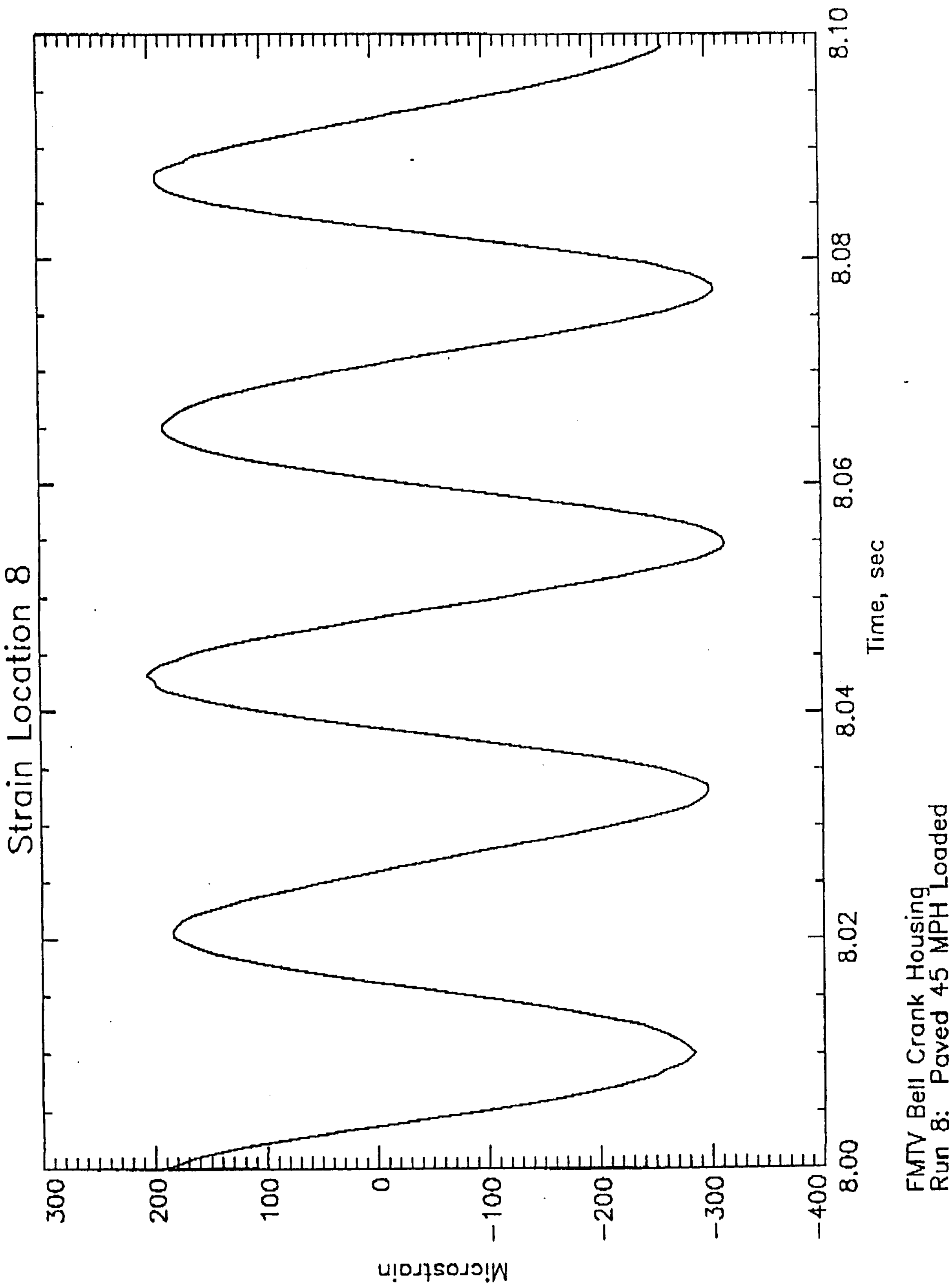
APG

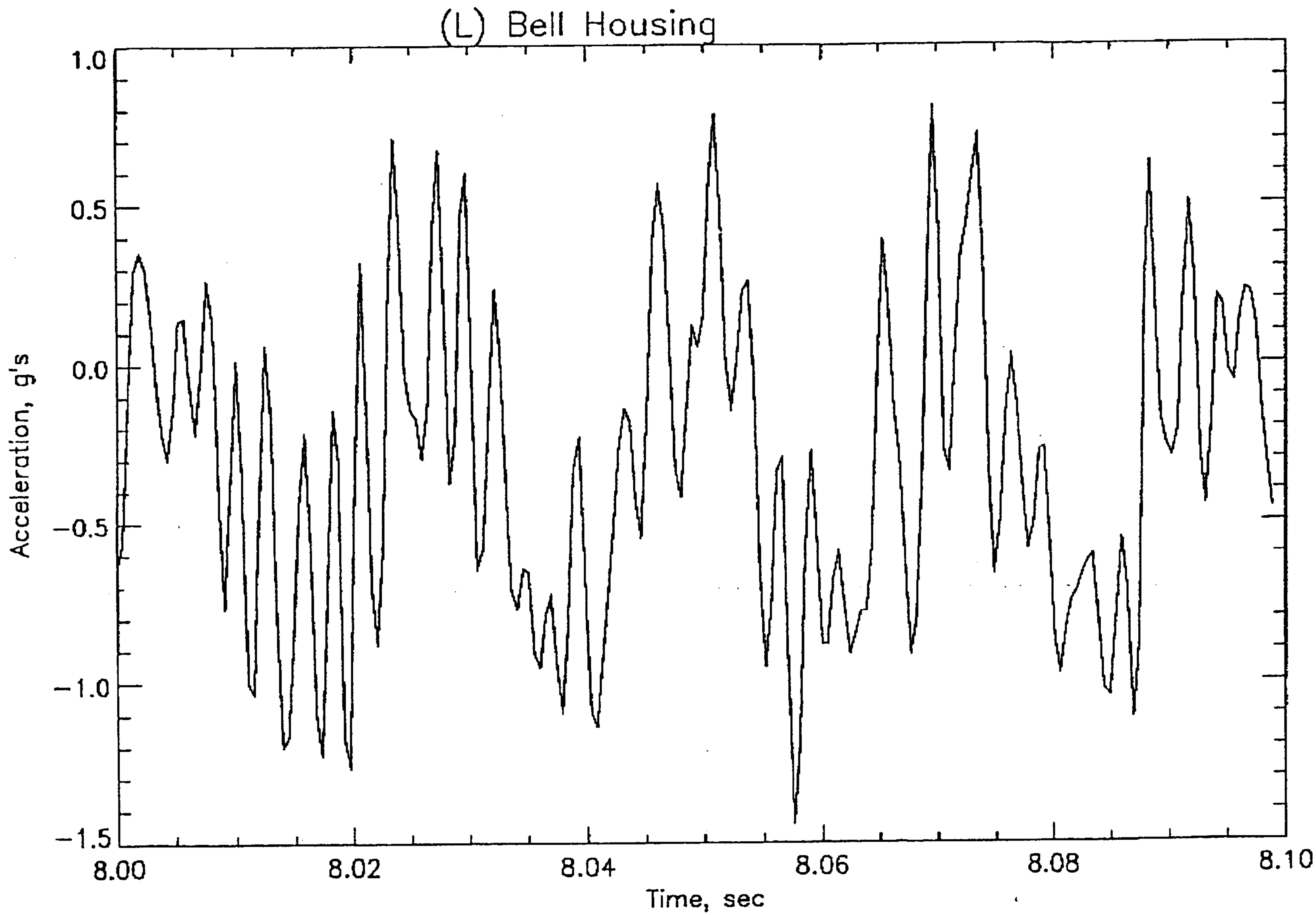


FMTV Bell Housing

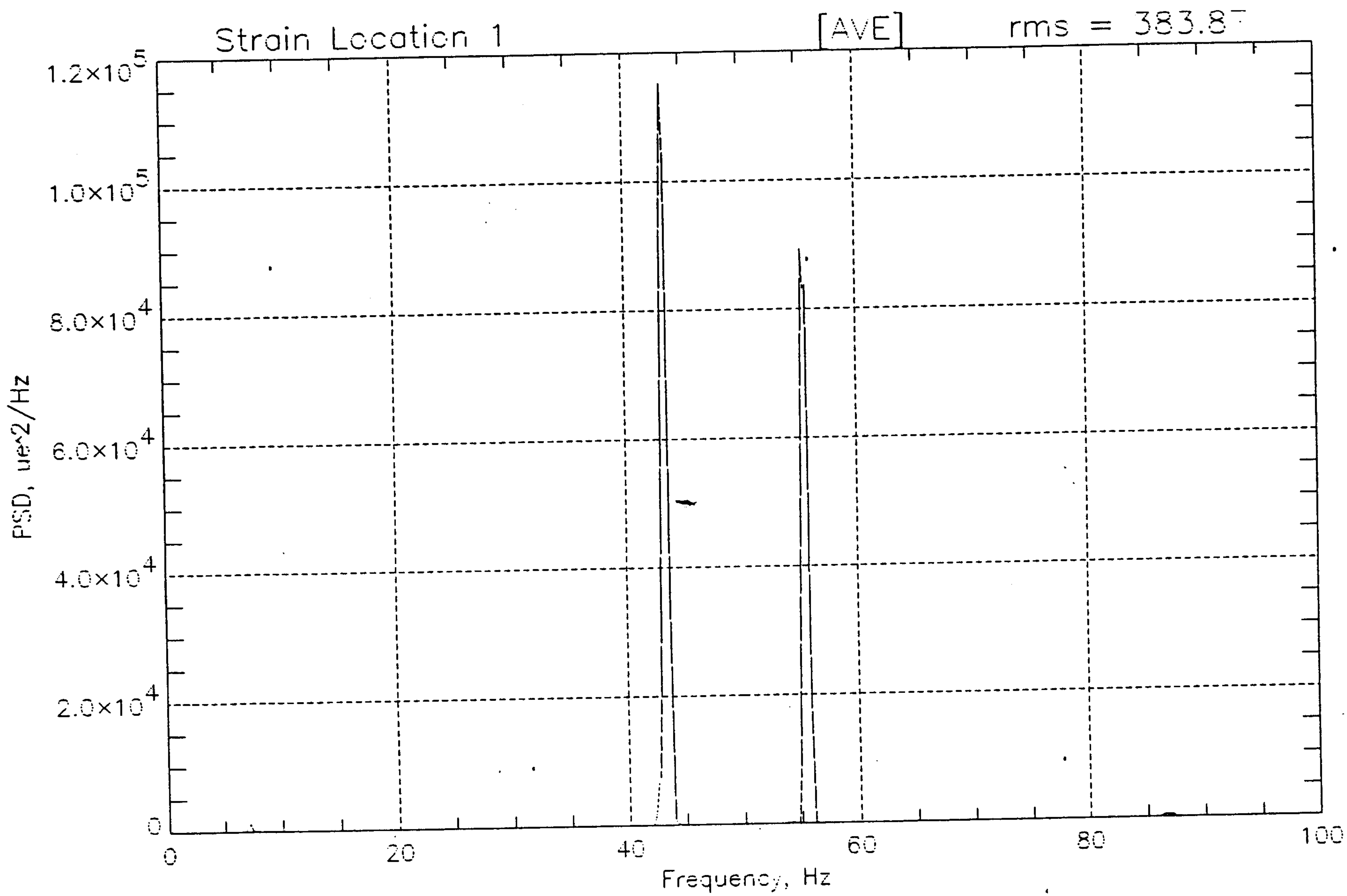


Strain Location 1
Microstrain
600
400
200
0
-200
-400
-600
Time, sec
8.00
8.02
8.04
8.06
8.08
8.10
FMTV Bell Crank Housing
Run 8: Paved 45 MPH Loaded

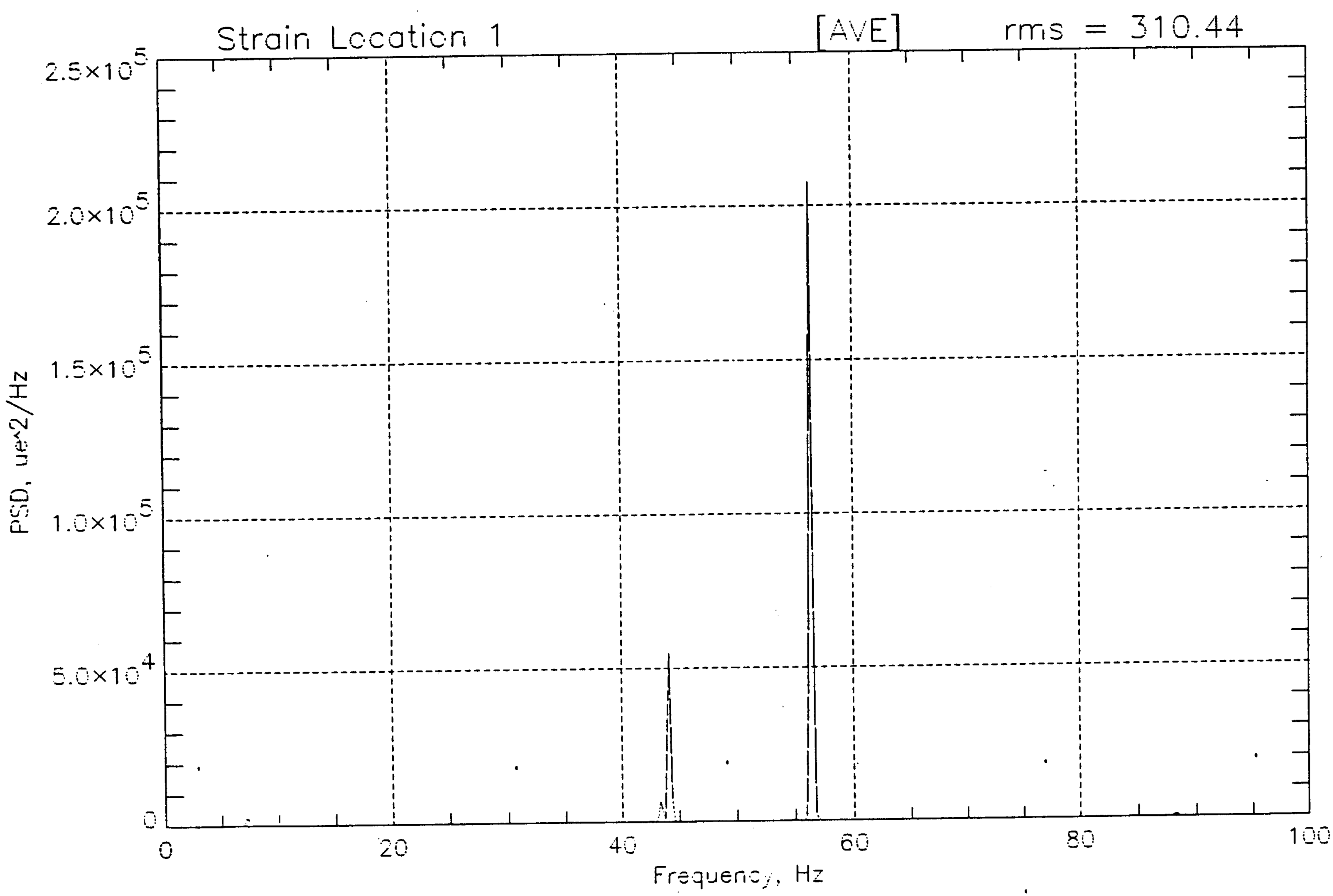

Strain Location 8
Microstrain
300
200
100
0
−100
−200
−300
−400
8.00
8.02
8.04
8.06
8.08
8.10
Time, sec
FMTV Bell Crank Housing
Run 8: Paved 45 MPH Loaded



FMTV Bell Housing
Run 8: Paved 45 MPH Loaded

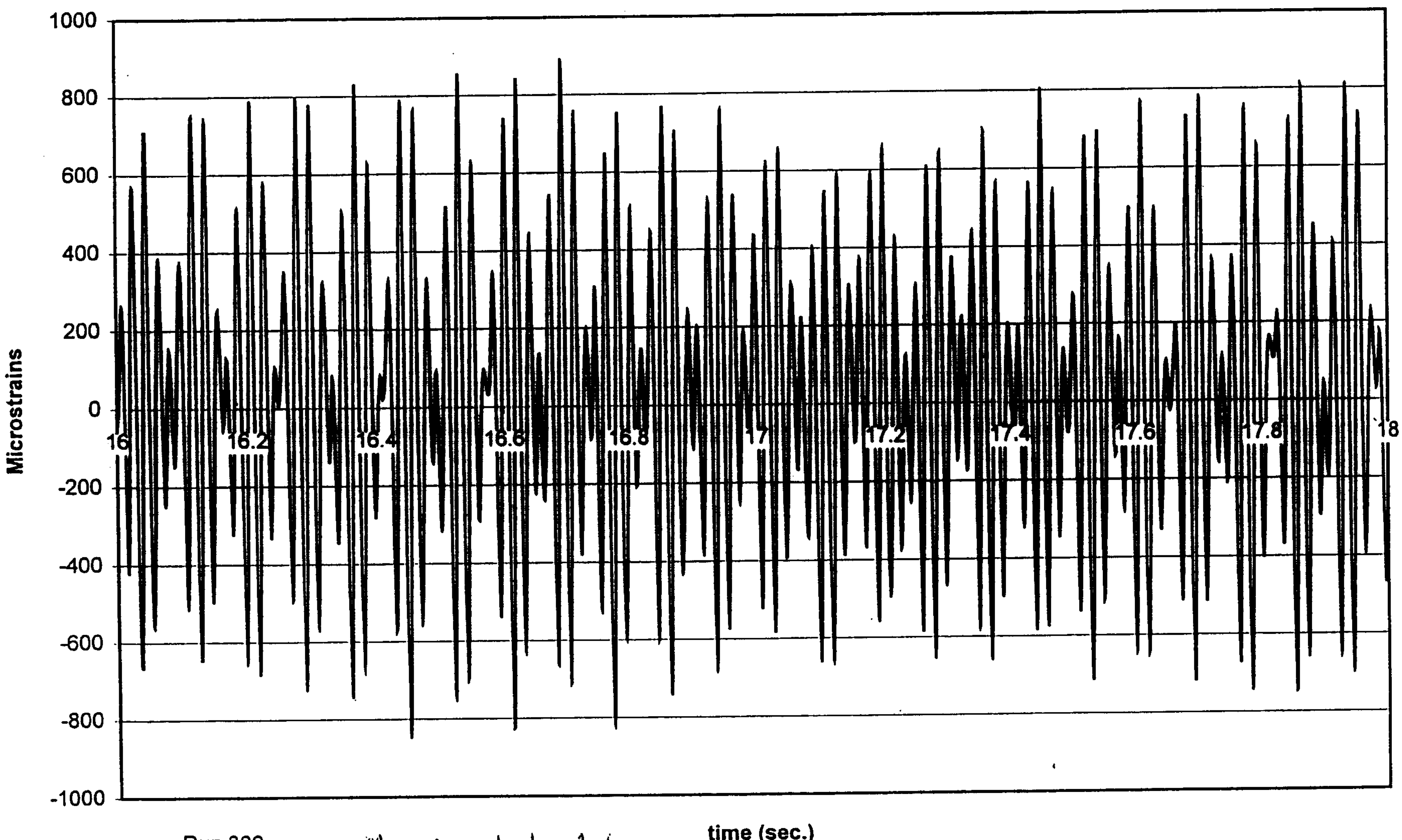


RUN 309: Paved, max mph, engine fan on, loaded
FMTV Bell Crank Housing
6 Averages; Delta f = 0.20 Hz; Block size = 2048; Hanning Window Enabled



RUN 303: Paved, max mph, engine fan off, loaded
FMTV Bell Crank Housing
6 Averages; Delta f = 0.20 Hz; Block size = 2048; Hanning Window Enabled

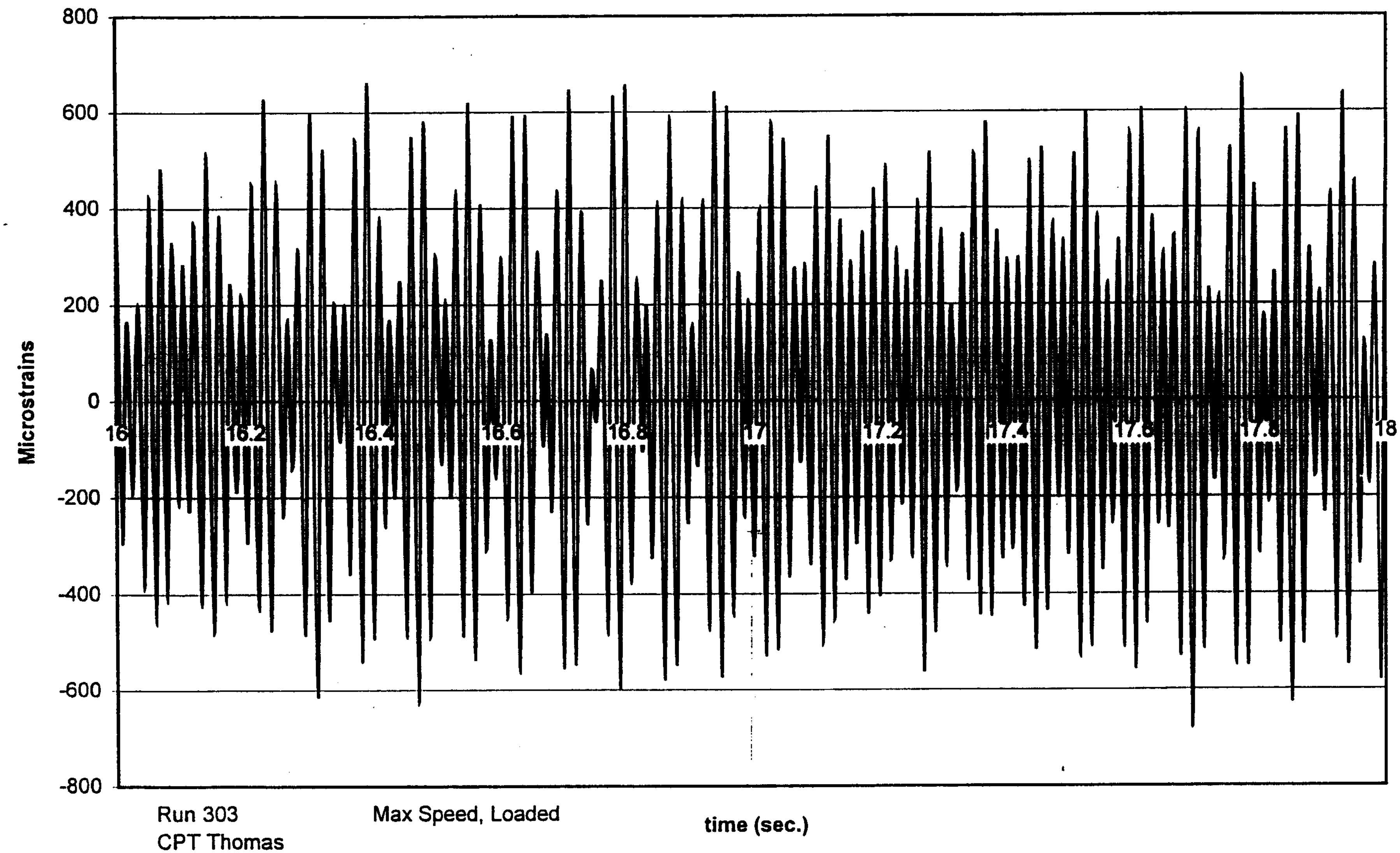
Engine Fan Engaged



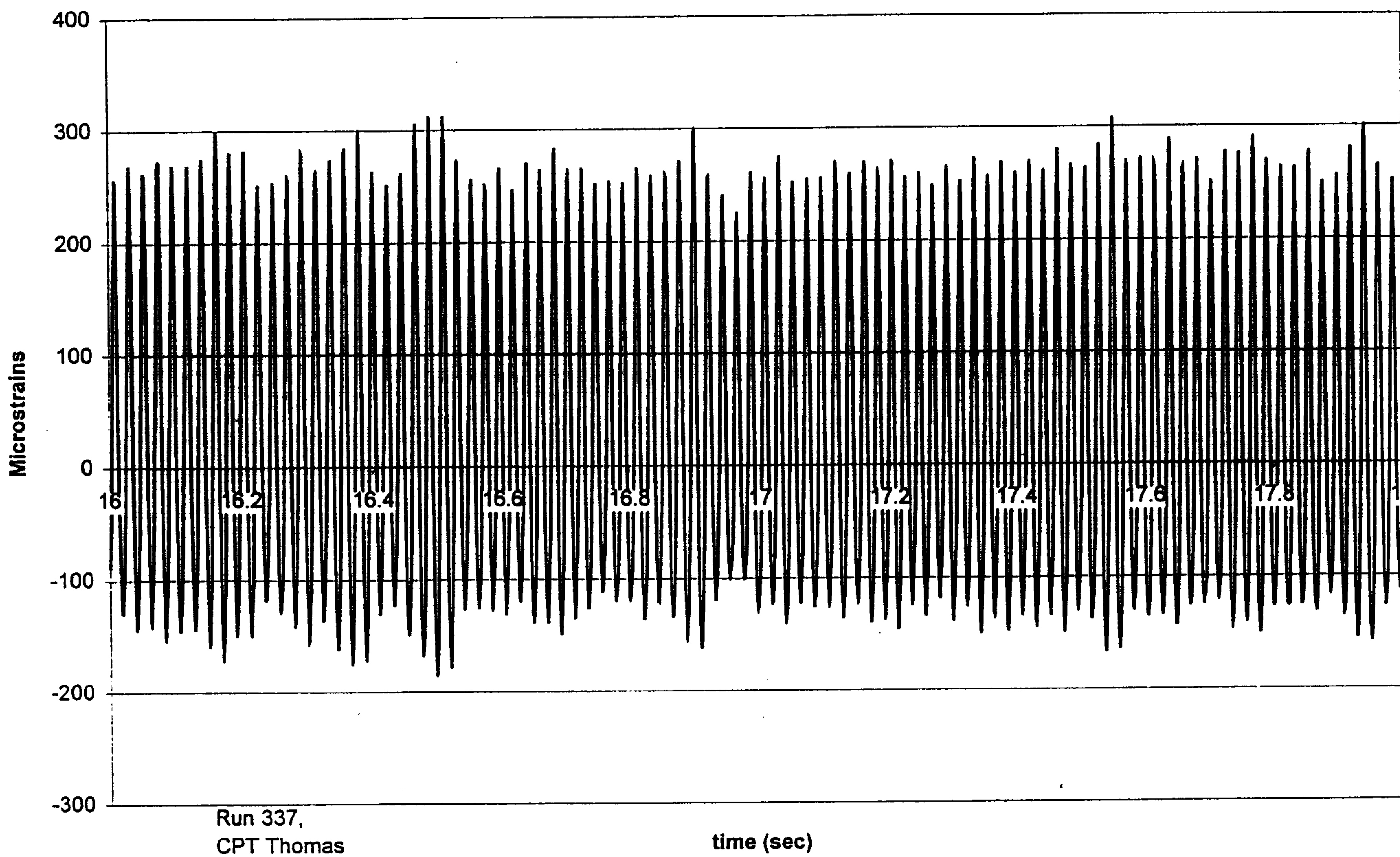
Run 309,
CPT Thomas

Max speed, Loaded

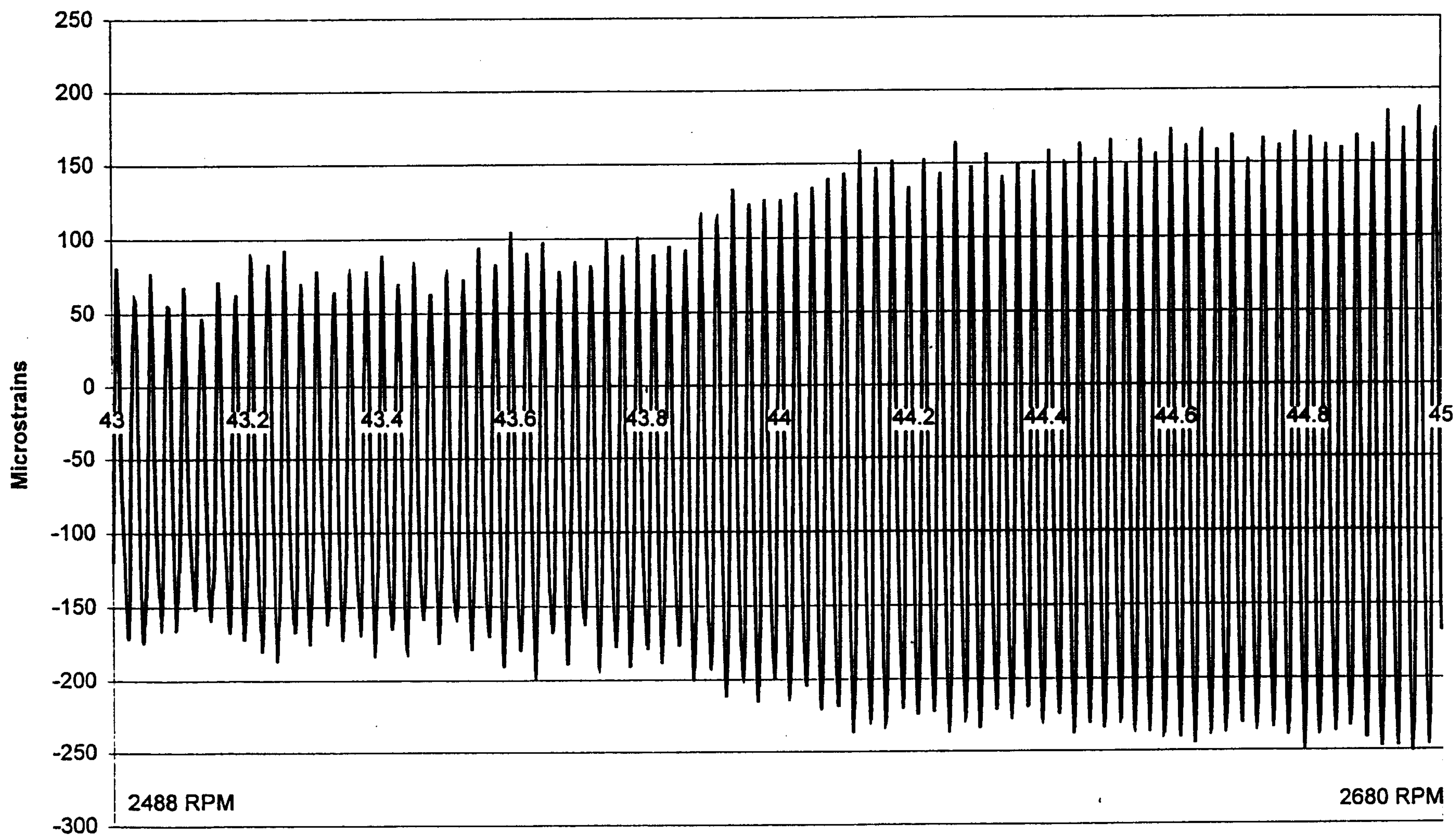
Engine Fan Disengaged



Max RPM Static, NO Engine Fan

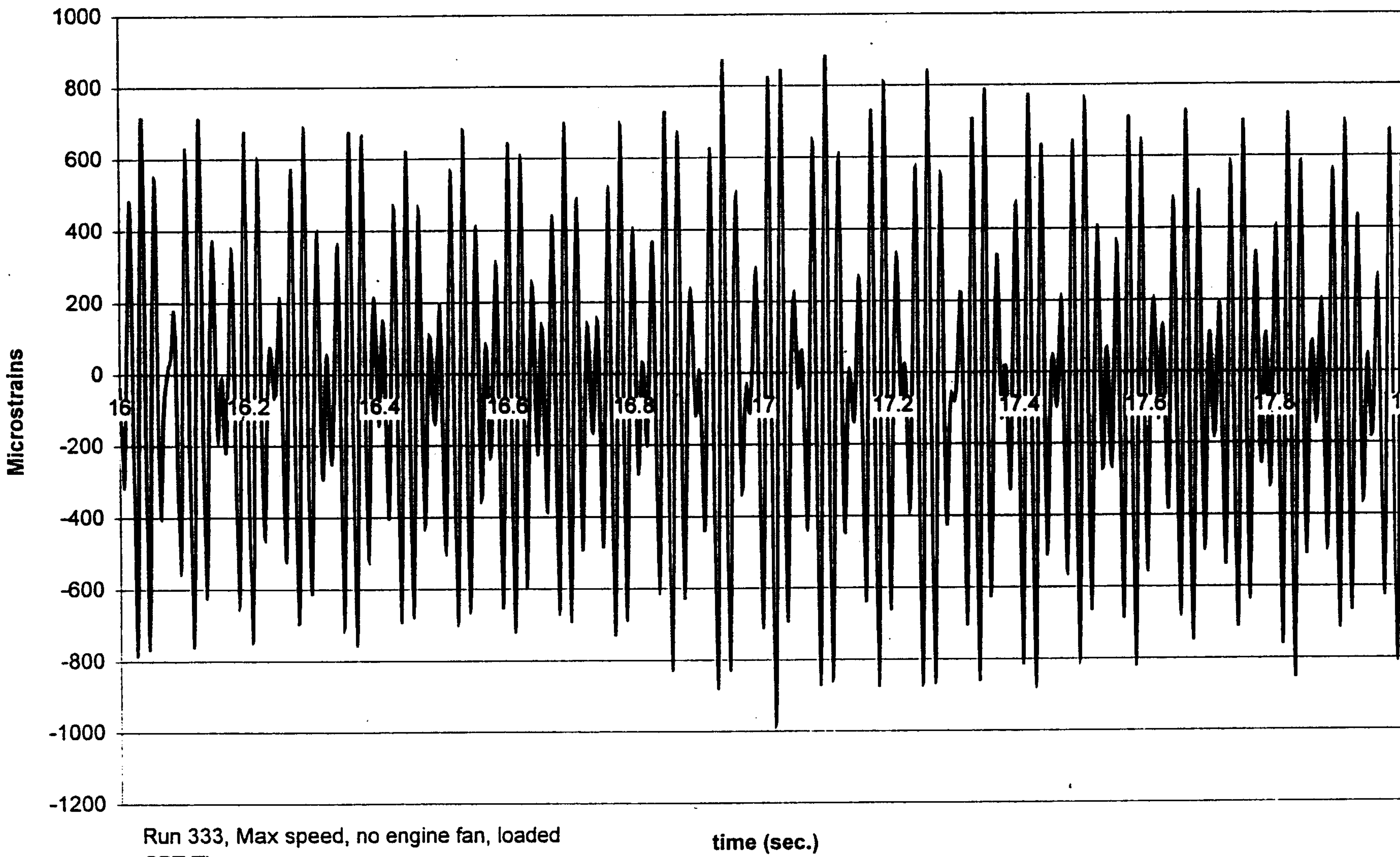


Rev to Max RPM



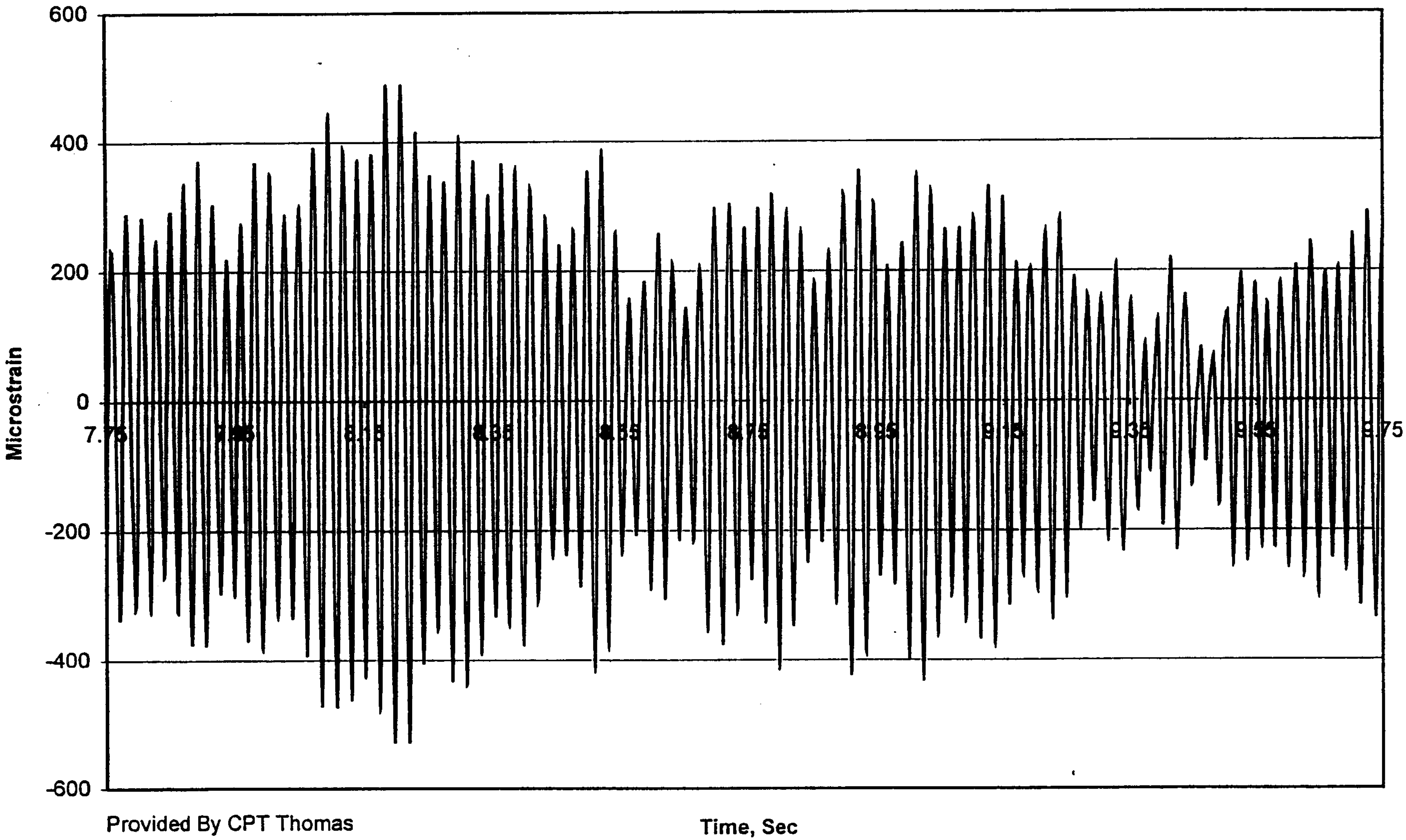
Run 305, Stationary rev to max RPM, Fan turned off
CPT Thomas

Max Speed NO Engine fan

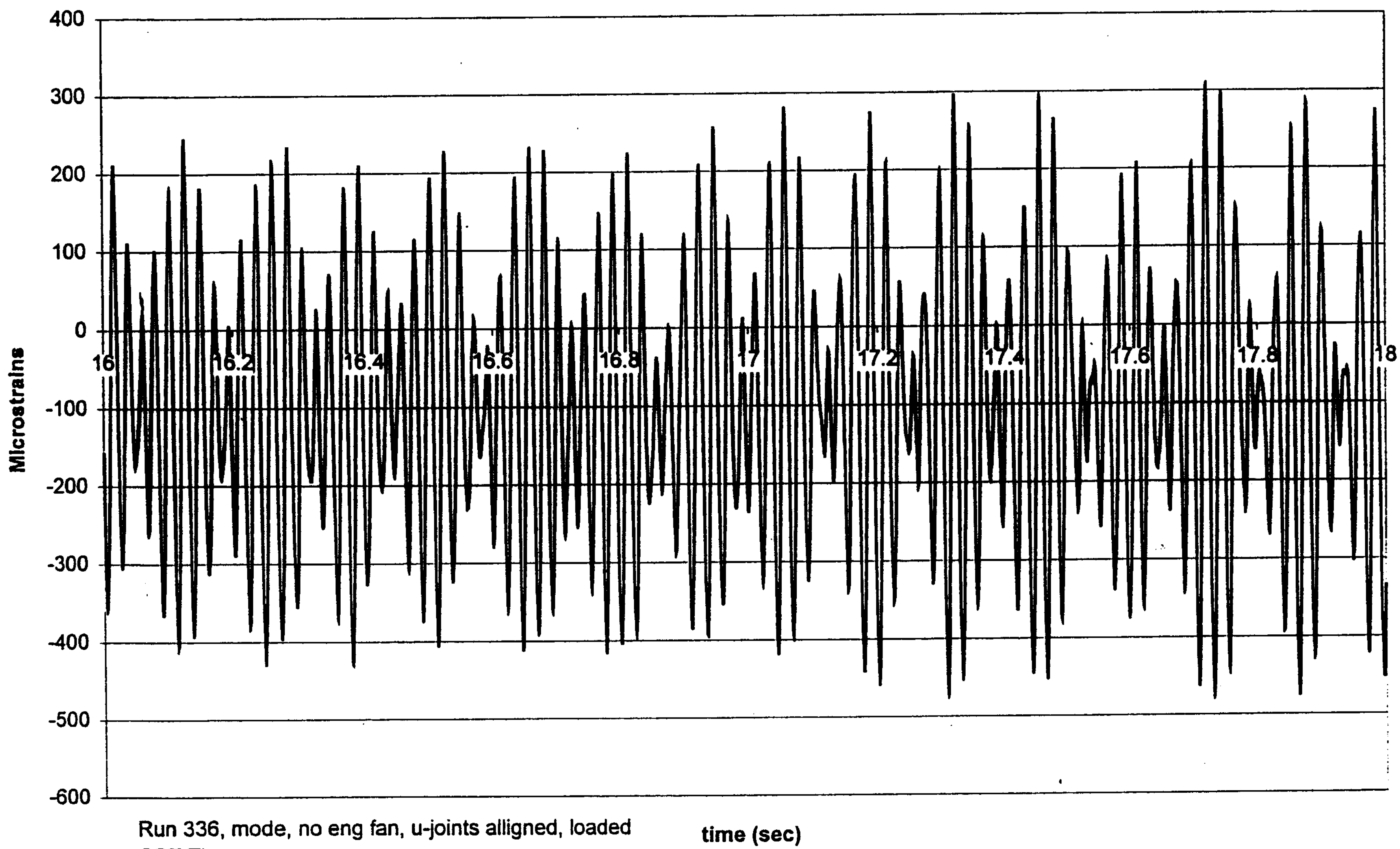


Run 333, Max speed, no engine fan, loaded
CPT Thomas

SG # 1, Run 8 45 MPH Paved , Loaded



37 MPH, in Mode, NO eng Fan



Run 336, mode, no eng fan, u-joints alligned, loaded
CPT Thomas

Plan of Attack:

1. Determine the state of well being of the test vehicle.
    a. Strain gage the flywheel housing
    b. Install accelerometers
    c. Record strain levels over the speed range of the vehicle
    d. Compare with other investigations

2. Separate the exciters
    a. Run known unbalances at the transfer case yokes. In and out of phase
    b. Run known unbalances at front of the engine (fan)
    c. Run with torque converter removed
    d. Draw conclusions – rank order critical components and limits

3. Simultaneously fabricate a stiffened stronger retuned system.

4. Design and fabricate a dynamic absorber to cancel the exciting forces from 40-60 Hz.

5. Test and conclude.

FIRST MODE VERTICAL BENDING @ 47.6 HZ

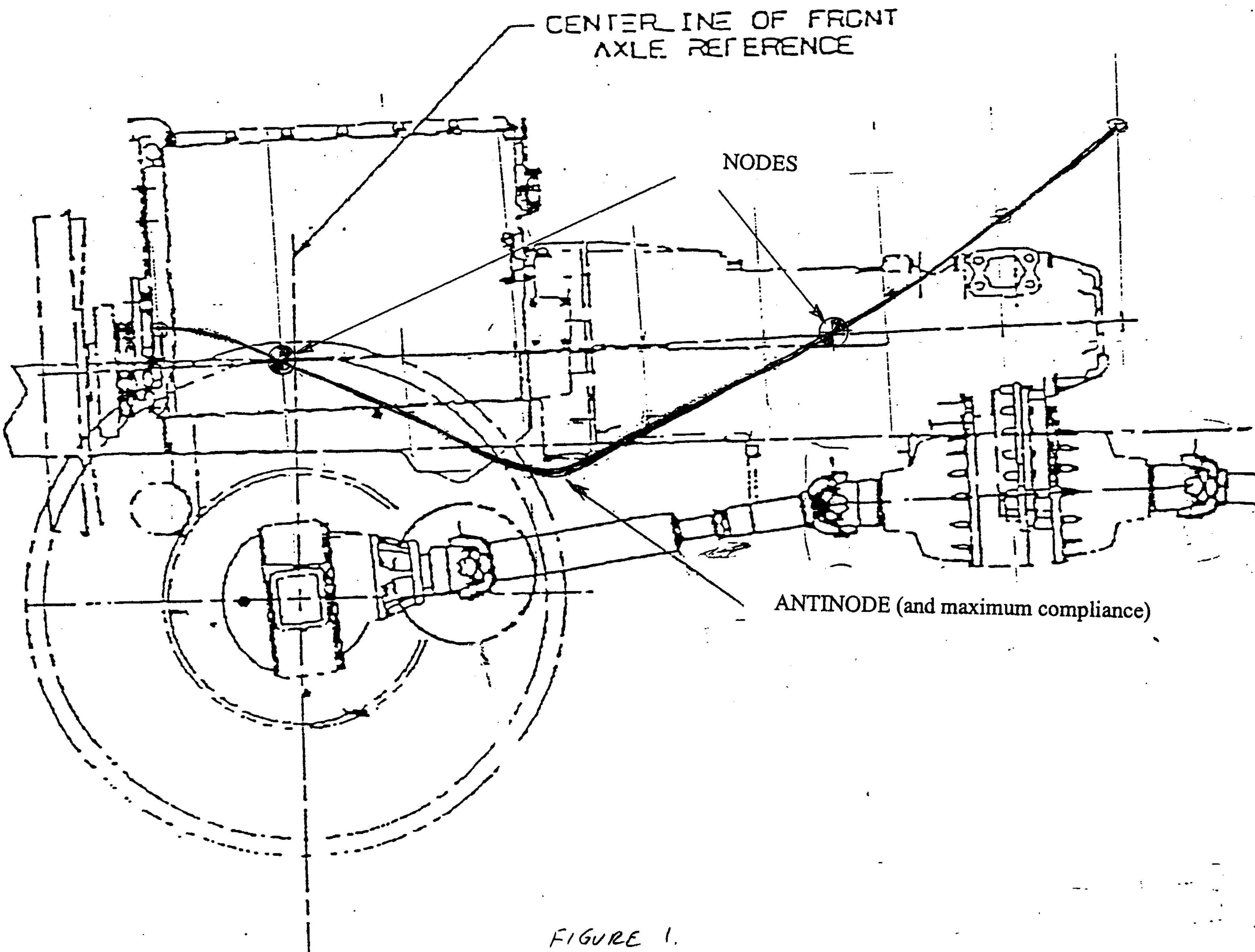


FIGURE 1.